High-Performance Digital Hydraulic Tracking Control of a Mobile Boom Mockup

Adj. Prof. Matti Linjama, Dr.-Ing. Mikko Huova, M. Sc. Otso Karhu & Prof. Kalevi Huhtala

Department of Intelligent Hydraulics and Automation (IHA), Tampere University of Technology, P.O.Box 589, 33101 Tampere, Finland, E-mail: matti.linjama@tut.fi

Abstract

The automation of hydraulic mobile machinery, such as excavators, requires high performance control solutions. In hydraulics, this means fast and accurate force, velocity and position control of hydraulic cylinder. Especially the force control is known to be difficult with traditional servo valves. Fast digital hydraulic valves together with modern control solutions can overcome this problem. This paper uses a new force control solution, which is based on the fast digital hydraulic valves and model based control principle. The control solution is applied in a heavy axis mimicking dynamics of mobile machine booms. Experimental results show good force, velocity and position tracking performance with varying load masses. The slow velocity performance is also much improved when compared to the earlier results.

KEYWORDS: Digital hydraulics, tracking control, force control, position control

1. Introduction

Hydraulics is used in many applications, such as aircraft actuators, paper mills and working machines, because of their high power-to-weight ratio. The increasing automation level sets new demands on the controllability of actuators. Accurate force, velocity and position tracking control are the basic functions of modern hydraulic actuators and several different solutions have been suggested. Kim et al. /1/ studied the flatness based non-linear control and demonstrated 0.3 mm position tracking error with the peak velocity of 63 mm/s. The commonly used performance index–the maximum error divided by the peak velocity–was thus about 0.005 s. Koivumäki and Mattila /2/ used virtual decomposition control in a construction crane and achieved performance index of 0.0030 s. Recently, Linjama et al. /3/ presented a model-based force, velocity and position tracking control solution for high inertia system. Fast digital hydraulic valves were used and simulated results with 500 kg load mass showed performance index of 0.0025 s. The benefits of the solution were simple controller structure, robustness against variations in bulk modulus, system delay and load mass, and no need for differentiation of velocity or pressures. This paper experimentally validates the results of /3/ in a heavily loaded 1-DOF boom. The target application is mobile machine and special attention is paid on the robustness against variation in the inertial load. Experimental results show good tracking performance with inertia between 10450 and 53000 kg. The performance index of 0.0095 s is achieved with the nominal load mass and 0.034 s with the full range of load mass variation.

2. Model Based Control of Asymmetric Cylinder

2.1. System Model

The controller is presented in detail in /3/ and only the main points are repeated here. The dynamics of a hydraulic cylinder with the inertia m and external force F_{load} can be expressed as:

$$\begin{aligned} \dot{p}_A &= \frac{B_A}{A_A x + V_{0A}}\left(Q_{PA} - Q_{AT} - A_A \dot{x}\right),\ \dot{p}_B = \frac{B_B}{A_B\left(x_{max} - x\right) + V_{0B}}\left(Q_{PB} - Q_{BT} + A_B \dot{x}\right) \\ m\ddot{x} &= p_A A_A - p_B A_B - F_\mu\left(\dot{x}\right) - F_{load} \end{aligned} \tag{1}$$

The flow rates Q_{PA}, Q_{AT}, Q_{PB}, and Q_{BT} are controlled by N parallel connected on/off valves and their flow rates are modelled by generalized turbulent flow model /4/:

$$Q_{XY} = \sum_{i=1}^{N} \mathbf{u}_{\mathbf{XY}}(i)\mathbf{K}_{\mathbf{v,XY}}(i)\operatorname{sgn}\left(p_X - p_Y\right)\left(\left|p_X - p_Y\right|\right)^{\mathbf{x}_{\mathbf{XY}}(i)} \tag{2}$$

where XY is either PA, AT, PB, or BT and •(i) refers to the i:th element of the vector. Combining equations 1 and 2 gives three coupled non-linear differential equations:

$$\begin{aligned} \dot{p}_A &= f_A\left(p_A, x, \dot{x}, \mathbf{u}_{\mathbf{PA}}, \mathbf{u}_{\mathbf{AT}}, p_P, p_T\right),\ \dot{p}_B = f_B\left(p_B, x, \dot{x}, \mathbf{u}_{\mathbf{PB}}, \mathbf{u}_{\mathbf{BT}}, p_P, p_T\right) \\ \ddot{x} &= f_x\left(p_A, p_B, \dot{x}, F_{load}\right) \end{aligned} \tag{3}$$

Control inputs of the system are control vectors of valve series ($\mathbf{u}_{\mathbf{PA}}$, $\mathbf{u}_{\mathbf{AT}}$, $\mathbf{u}_{\mathbf{PB}}$, $\mathbf{u}_{\mathbf{BT}}$), and uncontrollable inputs are supply pressure p_P, tank pressure p_T and load force F_{load}. The state variables are chamber pressures p_A and p_B, and piston position x and velocity $\dot{x}$.

2.2. Pressure and Force Controller

The fundamental assumption made is that the inertia m is so large that velocity does not change significantly during the one controller sampling period. This decouples the differential equations of Eq. 3 and makes it possible to estimate pressures using e.g. Heun’s method. The chamber pressures can then be controlled by selecting such control signals that the differences between the target pressures and the estimated pressures are minimized. The output of the hydraulic cylinder is piston force, which is given by:

$$F = p_A A_A - p_B A_B - F_\mu \tag{4}$$

Thus, there are two control variables and one output. One option is that the pressure differentials over the DFCUs are the same. This gives:

$$\begin{aligned} p_{A,ref} &= \frac{F_{ref}}{A_A + A_B} + \frac{A_B}{A_A + A_B}\left(p_P + p_T\right) \\ p_{B,ref} &= \frac{-F_{ref}}{A_A + A_B} + \frac{A_A}{A_A + A_B}\left(p_P + p_T\right) \end{aligned} \tag{5}$$

The primary objective of the force controller is to minimize the force error while the secondary objective is to keep pressures near the target values. This is solved by first selecting such control candidates for the A- and B-side that the estimated error in chamber pressures is small, and then calculating force error for each combination of the control candidates. The control candidates are selected according to the weighted sum of the estimated pressure error and the magnitude of the openings of the DFCUs:

$$\begin{aligned} J_A &= \left|p_{A,ref} - \hat{p}_A\left(k+1\right)\right| + W_{u,p}\mathbf{b}^{\mathrm{T}}\left(\mathbf{u_{PA}} + \mathbf{u_{AT}}\right) \\ J_B &= \left|p_{B,ref} - \hat{p}_B\left(k+1\right)\right| + W_{u,p}\mathbf{b}^{\mathrm{T}}\left(\mathbf{u_{PB}} + \mathbf{u_{BT}}\right) \end{aligned} \tag{6}$$

where $W_{u,p}$ is the weight for the opening of the DFCUs and the vector $\mathbf{b}$ determines the relative sizes of the valves. The N_{cand} candidates that give the smallest value for J_A and J_B are selected for further analysis. The final selection is made by minimizing the following penalty function:

$$\begin{aligned} J = &\left|F_{ref} - \hat{F}\left(k+1\right)\right| + W_{u,F}\mathbf{b}^{\mathrm{T}}\left(\mathbf{u_{PA}} + \mathbf{u_{AT}} + \mathbf{u_{PB}} + \mathbf{u_{BT}}\right) + \\ &W_{\Delta u,F}\mathbf{b}^{\mathrm{T}}\left(\left|\Delta\mathbf{u_{PA}}\right| + \left|\Delta\mathbf{u_{AT}}\right| + \left|\Delta\mathbf{u_{PB}}\right| + \left|\Delta\mathbf{u_{BT}}\right|\right) \end{aligned} \tag{7}$$

2.3. Pressure and Force Dynamics

It is shown in /3/ that the pressures have first order dynamics with time constant:

$$\tau = -\frac{T_S}{\ln\left(1 - B/\hat{B}\right)} \tag{8}$$

The equation is valid if the real bulk modulus B is smaller than the estimate of the bulk modulus $\hat{B}$. The A-side and B-side bulk moduli are different and the time constants of the A-pressure and B-pressure are different. If the supply and tank pressures are constant, the force dynamics can be expressed:

$$F(s)=\frac{\frac{\tau_B A_A+\tau_A A_B}{A_A+A_B}s+1}{(\tau_A s+1)(\tau_B s+1)}F_{ref}\mathrel{\hat{=}}G_F(s)F_{ref} \tag{9}$$

2.4. The Outer-Loop Velocity Controller

Equation 9 shows that, the inner-loop controller has the dynamics $G_F(s)$ with unknown time constants τ_A and τ_B. The upper limit of the time constants can be determined by estimating the lowest possible bulk modulus. The lower limit for the time constants is zero. In addition to time constants, the system has also delay d caused by the valves, the sampling, the computation, and the pipeline dynamics. Furthermore, the velocity is differentiated from the low-pass filtered position. The nominal open-loop transfer function of the system is selected to be:

$$G_N(s)=\frac{1}{m_{min}s}e^{-d_{min}s}G_{LP}(s) \tag{10}$$

where $G_{LP}(s)$ is the transfer function of the low-pass filter. The true transfer function is:

$$G_T(s)=\frac{1}{m\,s}G_F(s)e^{-ds}G_{LP}(s) \tag{11}$$

The multiplicative error caused by parameter variations is

$$\Delta(s)=\frac{G_T(s)-G_N(s)}{G_N(s)} \tag{12}$$

Now the system remains stable if $G_T(s)$ and $G_N(s)$ have the same number of unstable poles and following holds /5/:

$$|\Delta(j\omega)|<\left|\frac{1+G_N(j\omega)G_V(j\omega)}{G_N(j\omega)G_V(j\omega)}\right|\ \forall\omega>0 \tag{13}$$

Both $G_T(s)$ and $G_N(s)$ have one pole at origin and other poles are stable. Therefore, the stability criterion of Eq. 13 can be used. In practice, the robust tuning is made such that there is certain margin in Eq. 13 and that this margin increases with frequency. This takes into account the unmodelled dynamics, such as pipeline dynamics. One possible velocity controller is P-controller /3/:

$$G_V(s) = K_{P,vel} \tag{14}$$

2.5. Position Tracking Controller

The position loop design is made similarly to the velocity controller. The nominal and true open-loop transfer functions are:

$$G_{N,pos} = \frac{G_N(s)G_V(s)}{s\left(1+G_N(s)G_V(s)\right)} \quad , \; G_{T,pos} = \frac{G_T(s)G_V(s)}{s\left(1+G_T(s)G_V(s)\right)} \tag{15}$$

The multiplicative modelling error is:

$$\Delta_{pos}(s) = \frac{G_{T,pos}(s) - G_{N,pos}(s)}{G_{N,pos}(s)} \tag{16}$$

and the stability criterion is:

$$\left|\Delta_{pos}(j\omega)\right| < \left|\frac{1+G_{N,pos}(j\omega)G_P(j\omega)}{G_{N,pos}(j\omega)G_P(j\omega)}\right| \forall \omega > 0 \tag{17}$$

A simple PI-controller is selected for the position loop:

$$G_P(s) = K_{P,pos} + \frac{K_{I,pos}}{s} \tag{18}$$

The block diagram of the control system is shown in Figure 1.

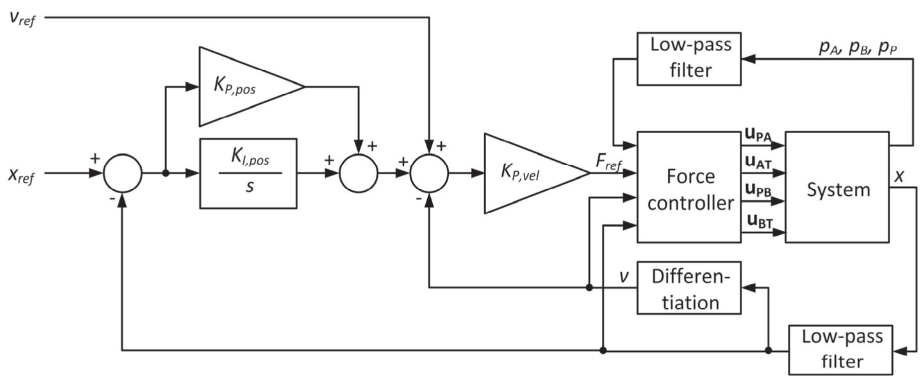

Figure 1. Block diagram of the controller.

3. Test System

The test system and its hydraulic circuit diagram are shown in Figure 2. The system mimics the dynamics of the joint actuator of a typical medium sized mobile machine boom and its natural frequency is about 3 Hz with 400 kg load mass. The pressures are measured with Druck PTX1400 sensors with 0-40 MPa measurement range and 4-20 mA current output, and the 500 Ohms precision resistor is used to obtain voltage signal. The piston position is measured by a potentiometer with $\pm$10 VDC supply voltage. The position signal has analogue RC filter with 0.22 ms time constant. The voltages are measured with dSPACE DS2002 16-bit ADC board. The controller is implemented by Simulink and dSPACE DS1006 controller board.

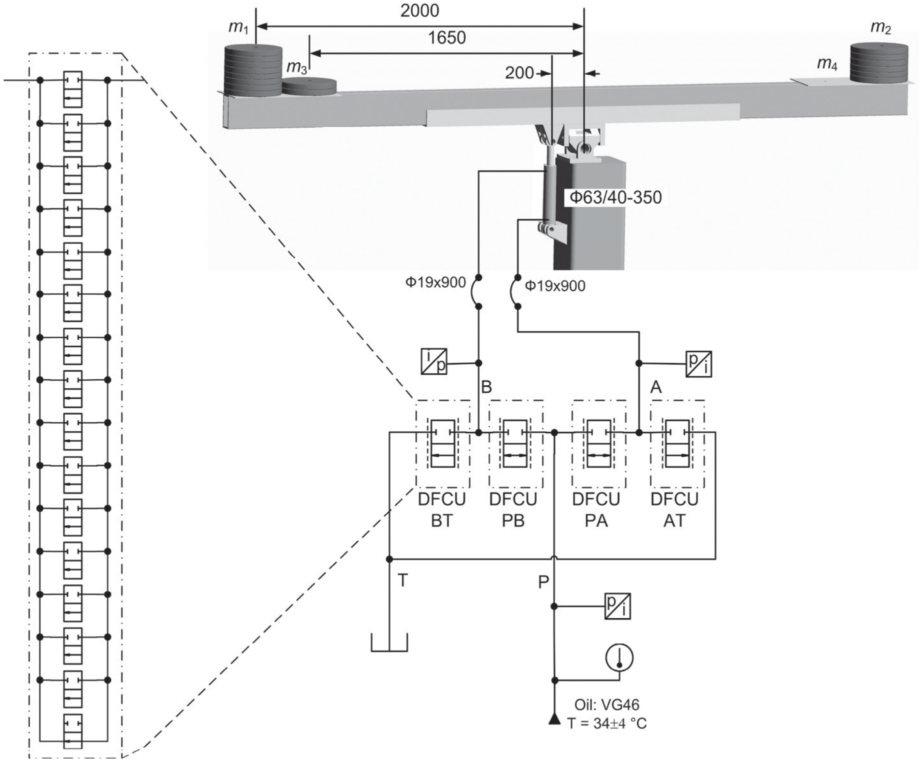

Figure 2. The system studied.

The digital valve system consists of 4$\times$16 miniaturized fast on/off valves /6/. All valves have the same flow capacity. The average valve parameters are identified from the measured responses having six valves open in DFCUs PA and BT. The measurements are repeated with 5, 6, 7, 8, 9 and 10 MPa supply pressure and the flow rate and corresponding pressure differentials are determined from the pressure and velocity measurements. The valve parameters are $K_{V,PAi} = K_{V,PBi} = 8.7\times10^{-9}$ m^3/(s Pa$^{0.53}$), $K_{V,ATi} = K_{V,BTi} = 7.5\times10^{-9}$ m^3/(s Pa$^{0.53}$) and exponent x is 0.53 for all valves.

4. Tuning of The Controllers

4.1. Force Controller

The parameters of the force controller are the sampling time T_S and weight factors $W_{u,p}$, $W_{u,F}$ and $W_{\Delta u,F}$. The base sampling period of the controller is selected to be 5 ms, i.e. slightly more than the maximum response time of the valves. The weight factors are selected based on simulations such that the simultaneous opening of DFCUs PA and AT, and DFCUs PB and BT as well as activity of valves are minimized but the force tracking performance is still good. The values used are $W_{u,p}$ = 0.05 MPa, $W_{u,F}$ = 10 N, and $W_{\Delta u,F}$ = 50 N. The estimate of the bulk modulus is selected to be 1500 MPa, which is considered as the biggest possible value. The parameter N_{cand} is selected to be eight.

4.2. Velocity and Position Controllers

The minimum inertia of the system is obtained without load masses and is estimated to be 418 kg m^2. The inertia reduced to the cylinder is thus 10450 kg at the horizontal orientation. The maximum inertia is obtained with 400 kg load mass and is estimated to be 53000 kg at the horizontal orientation. It is assumed that the true bulk modulus is at least 700 MPa, which results in maximum time constants of 8 ms. The maximum system delay is assumed to be the 3 ms valve delay plus the sampling period, i.e. 8 ms, and minimum delay is assumed to be 3 ms. The tuning of the velocity controller is made by plotting both sides of Eq. 13 with different parameter values. The seven equally spaced parameter values are used as follows: τ_A = 1–8 ms, τ_B = 1–8 ms, d = 3–8 ms, m = 10450–53000 kg. Figure 3 shows the both sides of Eq. 13 for the tuning $K_P = 0.45\times10^6$ Ns/m. Figure 3 shows also the closed-loop velocity step responses of the linear model. The position controller is tuned by plotting the both sides of Eq. 17 with different parameter values. The resulting tuning is $K_{P,pos}$ = 12 s^{-1} and $K_{I,pos}$ = 12 s^{-2}. Figure 3 shows the both sides of Eq. 17 for this tuning and corresponding step responses.

4.3. Implementation Issues

Pressure and position signals are filtered with a non-linear filter, which buffers five data points, removes the minimum and maximum value and takes the mean value of the remaining three data points. Pressures and position are then filtered with a first order linear filter. The time constants are 0.53 ms for chamber pressures, 5.3 ms for piston position and 8.0 ms for the supply pressure. The piston velocity is differentiated from the filtered piston position using filter suggested in /7/:

$$\hat{\dot{x}}(k) = \frac{5x(k)+3x(k-1)+x(k-2)-x(k-3)-3x(k-4)-5x(k-5)}{35T_S} \tag{19}$$

All filters run with 0.25 ms sampling period. The integrator term has ±0.1 mm deadzone in order to avoid limit cycles /3/.

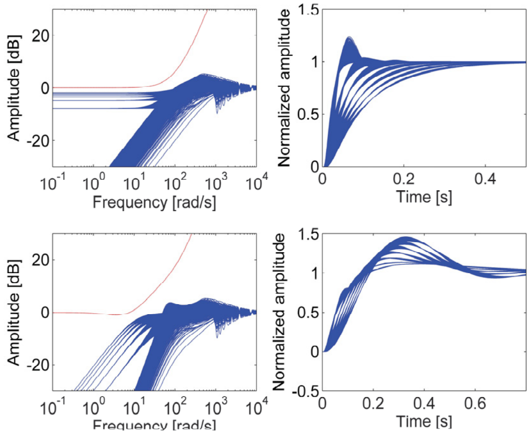

Figure 3. Both sides of Eq. 13 (top-left) and velocity step responses (top-right); both sides of Eq. 17 (bottom-left) and position step responses (bottom-right) when system parameters are varied.

5. Experimental Results

The supply pressure is 10 MPa in all measurements. Figure 4 presents the measured response without load masses. The position reference is the fifth order polynomial with 50 mm amplitude and 1.25 s movement time. The peak velocity is 75 mm/s and the maximum position tracking error is 0.71 mm. The performance index is thus 0.0095 s. The figure shows also that pressure tracking and force tracking is good but valves are quite active because of measurement noise. Figure 5 depicts the measured response when the boom has 200 kg load mass at both ends. The tracking error increases to 2.58 mm and the performance index increases to 0.034 s. The response is stable but there is small overshoot when stopping the movement as predicted by the linear model (see Figure 3). Figure 6 shows the measured slow velocity response 200 + 200 kg load mass. The amplitude of the movement is 5 mm and the peak velocity is 7.5 mm/s. The response shows that the system is capable for very slow motion.

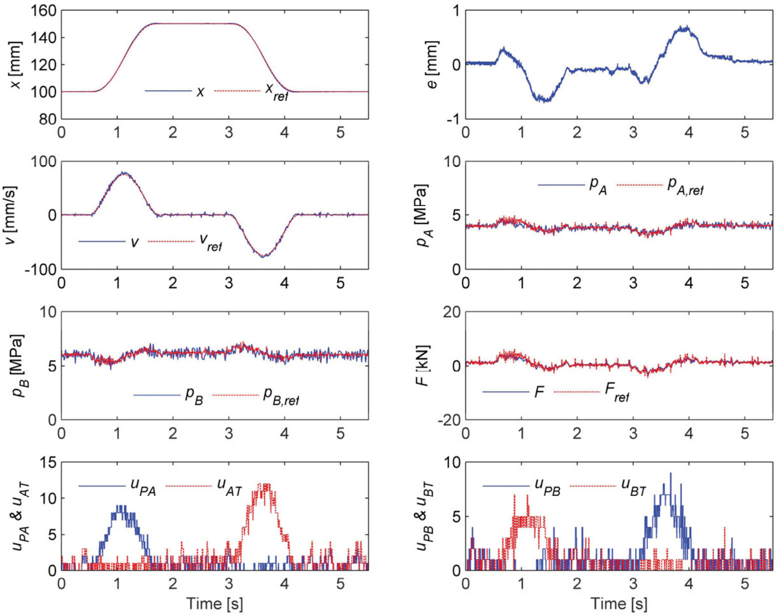

Figure 4. Position tracking results without load masses. Effective inertia is about 10450 kg.

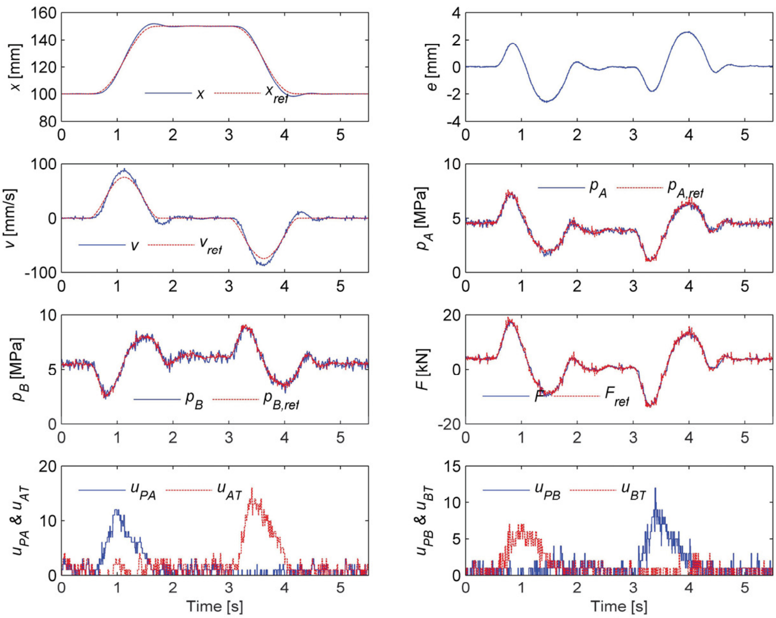

Figure 5. Position tracking results with 200 + 200 kg load masses. Effective inertia is about 53000 kg.

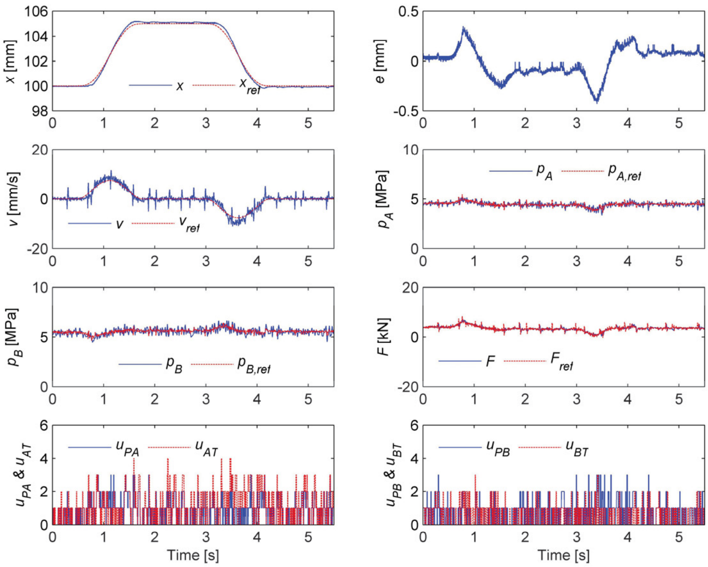

Figure 6. Small amplitude position tracking results with 200 + 200 kg load masses.

6. Conclusions

The new high performance digital hydraulic force, velocity and position tracking control solution has been experimentally validated. The solution is simple, does not need derivatives of velocity or pressure, and gives good control performance under greatly varying load mass and uncertain bulk modulus and system delay. The slow velocity performance is also good despite moderate resolution of the digital valve system. However, the results are not as good as simulations predicted in /3/. The reasons for this are larger variation in inertia and measurement noise. The measurement noise increases also activity of the valves.

Acknowledgement

The research was supported by Academy of Finland (Grant No. 286401).

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Nomenclature

A_A, A_B	Piston areas	m^2
b	Vector of relative sizes of valves of DFCUs	
B_A, B_B	Bulk moduli of the piston and rod side chambers	Pa
d, d_{min}	Actual and minimum system delay	s
e	Position error	m
F, F_{load}, F_{ref}	Actual force, load force, force reference	N
F_μ	Friction force	N
$G_N(s)$,	Nominal transfer function of the velocity loop	
$G_{N,pos}(s)$	Nominal transfer function of the position loop	
$G_P(s)$, $G_V(s)$	Transfer function of the position and velocity controller	
$G_T(s)$	True transfer function of the velocity loop	

$G_{T,pos}(s)$	True transfer function of the position loop	
J	Cost for the force error	N
J_A, J_B	Cost for the piston side and rod side pressure error	Pa
$K_{I,pos}$, $K_{P,pos}$	I-term and P-term gain of the position controller	$1/s^2$
$K_{P,vel}$	Velocity controller gain	Ns/m
m, m_{min}	Actual and minimum load mass	kg
N	Number of parallel connected valves	
p_A, p_B	Pressure in the piston side and rod side chamber	Pa
$p_{A,ref}$, $p_{B,ref}$	Reference value for the piston side and rod side pressure	Pa
p_P, p_T	Supply and return line pressure	Pa
Q_{XY}	Flow rate of the valve system from port X to port Y, XY is either PA, AT, PB or BT	m^3/s
T_S	Sampling period of the controller	s
$\mathbf{u}_{\mathbf{XY}}$	Vector of the control signals of the DFCU XY, where XY is either PA, AT, PB or BT	
v, v_{ref}	Piston velocity and velocity reference	m/s
V_{0A}, V_{0B}	Dead volumes of the piston side and rod side chamber	m^3
$W_{u,F}$, $W_{\Delta u,F}$	Weight for the opening and change of opening of DFCUs	N
$W_{u,p}$	Weight for the opening of DFCUs	Pa
x, x_{max}, x_{ref}	Actual, maximum and reference piston position	m
$\Delta(s)$, $\Delta_{pos}(s)$	Modelling error of the velocity and position loop	
τ_A, τ_B	Time constants of pressure dynamics	s
$\hat{\bullet}$	Estimate of $\bullet$	